CRYOGENIC REFRIGERATOR

Patent number:	GB1483356	
Publication date:	1977-08-17	
Inventor:		
Applicant:	PHILIPS NV	
Classification:		
- international:	F25B9/00	
- european:	F02G1/044V; F25B9/14	
Application number:	GB19740039061 19740906	
Priority number(s):	NL19730012488 19730911	

Also published as:

- NL7312488 (A)
- JP50055951 (A)
- FR2243401 (A1)
- DE2442556 (A1)
- CH574588 (A5)

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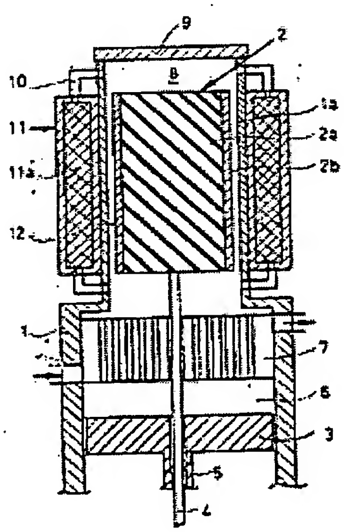
Abstract of **GB1483356**

1483356 Refrigerating PHILIPS GLOEI-LAMPENFABRIEKEN NV 6 Sept 1974 [11 Sept 1973] 39061/74 Heading F4H In a cryogenic refrigerator having a displacer 2 and piston 3 arranged to reciprocate with a phase difference and defining a compression space 6 and expansion space 8, a cooling unit 7 and a main regenerator 11 incorporated in a duct 10 and containing for example, lead balls; an auxiliary regenerator is connected in parallel with the main regenerator 11 and is formed by the annular gap between the displacer 2 and the cylinder 1a. The walls 2b of the displacer and the cylinder wall la are of stainless steel. The working medium is preferably helium. The hydraulic diameter d h of the annular gap satis- fies the relation: where s=stroke length of the displacer 2; #= mean dynamic viscosity of the working medium flowing through the gap during operation of the refrigerator; L = length of the gap in the axial direction; #=mean density of the working medium flowing through the gap during opera- tion of the refrigerator; ##=mean pressure drop through the main regenerator 11.

$$0{\cdot}4\times 2{\cdot}8\sqrt[4]{\frac{s.\eta^2.L}{P.\Delta P}} \leqslant d_L \geqslant 1{\cdot}4\times 2{\cdot}8\sqrt[4]{\frac{s\cdot\eta^2.L}{P.\Delta P}}$$



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PATENT SPECIFICATION

(11) 1 483 356

(21) Application No. 39061/74 (22) Filed 6 Sept. 1974
(31) Convention Application No. 7312488
(32) Filed 11 Sept. 1973 in
(33) Netherlands (NL)
(44) Complete Specification published 17 Aug. 1977
(51) INT CL² F25B 9/00
(52) Index at acceptance
F4H G3G G3H

(19)

(54) CYROGENIC REFRIGERATOR

(71) We, N. V. PHILIPS' GLOEILAMPENFABRIEKEN, a limited liability Company, organized and established under the laws of the Kingdom of the Netherlands, of Emmasingel 29, Eindhoven, Holland, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

The invention relates to a cryogenic refrigerator of the kind (hereinafter referred to as the "kind described") comprising a compression space of variable volume communicating with an expansion space of variable volume which has a lower mean temperature than the compression space during operation of the refrigerator, a displacer which is reciprocable in a cylinder to vary the volume of the expansion space, and a regenerator arranged in the communication between the said spaces, through which communication a gaseous working medium flows to and fro between said spaces during the operation of the refrigerator.

Refrigerators of the kind described are known.

Included in this kind of refrigerator are, for example, machines operating according to the Stirling cycle (United States Patent Specifications 2,907,175 and 3,400,544), machines operating according to the Vuilleumier cycle (United States Patent Specifications 1,275,507; 2,657,552 and 3,523,427) and machines of the Gifford-McMahon type (United States Patent Specifications 2,906,101 and 2,966,035).

In such machines the regenerator normally consists of a filling mass of gas-pervious material having a good heat capacity, for example, phosphor-bronze gauze layers or lead balls, contained in a housing. In order to minimize heat transfer by gas leakage from the compression space, which is at the higher temperature level, to the expansion space, which is at the lower temperature level, a seal, usually made of a synthetic resin material, is normally provided between the moving displacer and the cylinder wall. In addition to the cost of manufacture and fitting, this seal involves the drawback of, on the one hand, frictional losses and, on the other hand, wear causing gas leakage and also involving the risk of contamination of the regenerator by particles eroded from the seal by the wear.

It will be obvious that it would be very advantageous if the seal could be dispensed with. This has been done in the refrigerator described in our prior United Kingdom Patent Specification 1,335,854, in which the regenerator is exclusively formed by an annular gap between the displacer and the cylinder wall cooperating therewith.

This may be a solution for small refrigerators of low cooling power, in which the entire small flow of working medium can flow through the narrow gap from the compression space to the expansion space and *vice versa* substantially without flow loss and in proper thermal contact with the gap walls, but for large refrigerators of comparatively high cooling power, and hence involving comparatively large flows of working medium, this is a less attractive proposition. In the case of large flows of working medium, the flow losses and a poor regenerative action become too dominant.

According to the invention there is provided a cryogenic refrigeration of the kind described, wherein the communication between the compression and expansion spaces incorporates an auxiliary regenerator which is connected in parallel with the main regenerator and which is formed by an annular gap between the displacer and the cylinder wall cooperating therewith, at least one of the two surfaces of the displacer and said cylinder wall which face each other and bound the gap being formed of a material having a good thermal capacity with respect to the working medium flowing through the gap during operation of the refrigerator, and the hydraulic diameter d_h of the gap satisfying the relation:

$$0.4 \times 2.8 \sqrt[4]{\frac{s.\eta^2.L}{\rho.\Delta P}} \leqslant d_h \leqslant 1.4 \times 2.8 \sqrt[4]{\frac{s.\eta^2.L}{\rho.\Delta P}}$$

where

s=stroke length of the displacer,

η=mean dynamic viscosity of the working medium flowing through the gap during operation of the refrigerator,

L=length of the gap in the axial direction,

ΔP=mean pressure drop through the main regenerator,

ρ=mean density of the working medium flowing through the gap during operation of the refrigerator.

The hydraulic diameter of an annular gas duct, the annular gap in the present invention, is equal to four times the area of the cross-section of the duct divided by the sum of the lengths of the inner and outer boundaries of the cross-section.

The major difference between this refrigerator and the refrigerator known from our aforesaid Patent Specification 1,335,854 is that, whilst in this known machine the entire flow of working medium passes through the gap regenerator, in the present case there are two flows of working medium: a main flow through the main regenerator and a secondary flow through the auxiliary regenerator which is connected in parallel with the main regenerator. During operation of the refrigerator, flow losses in the main regenerator due to flow resistance therein cause a mean pressure drop ΔP through this regenerator. This pressure drop ΔP is transmitted to the ends of the auxiliary regenerator. In the known machine, in which the gap forms the only regenerator, this gap is not subjected to an external pressure difference but a pressure drop occurs in the gap because of flow losses. The above structural and physical differences in the present case require the hydraulic diameter of the gap to satisfy a relation which is very different from the relation given in the aforesaid Patent Specification 1,335,854.

The hydraulic diameter of the regenerative gap is equal to twice the gap width.

If the hydraulic diameter satisfies the relation given above, a good heat transfer from the secondary flow of working medium to the gap walls and *vice versa* is ensured, whilst the flow resistance in the gap is low.

An embodiment of the invention will now be described in detail with reference to the accompanying drawing, which is a diagrammatic axial sectional view of a refrigerator of the kind described operating according to the Stirling cycle (cold-gas refrigerator).

This refrigerator comprises a cylinder 1 in which a displacer 2 and a piston 3 are arranged to reciprocate with a phase difference. Displacer 2 compresses a cylindrical body 2a of a synthetic resin material of low thermal conductivity, surrounded by a thin stainless steel jacket 2b. The displacer 2 is connected to a drive mechanism (not shown) by way of a displacer rod 4, whilst the piston 3 is connected to the same mechanism by way of a hollow piston rod 5 through which the displacer rod 4 slidably passes. Between the piston and the displacer is a compression space 6 accommodating a cooling unit 7. Above the displacer 2 is an expansion space 8 which is bounded at its upper side by a freezer plate 9 which is a heat-exchanger *via* which heat can be extracted from an object to be cooled by means of the cold produced in the expansion space 8. The way in which the cold-gas refrigerator operates to produce this cold is well-known and need not be described herein.

Compression space 6 and expansion space 8 are in open communication with each other *via* a main regenerator 11 incorporated in a duct 10 and containing, for example, lead balls as a regenerative filling mass 11a, and *via* an auxiliary regenerator which is connected in parallel with the main regenerator and which is formed by an annular gap 12 between the stainless-steel jacket 2b of displacer 2 and the cylinder wall 1a cooperating therewith, which wall is also made of stainless steel.

The working space of the refrigerator, which comprises the compression and expansion spaces 6 and 8 and the communication between them, namely, the duct 10 and regenerators 11 and 12, contains a gaseous working medium, for example, helium.

On its way from compression space 6 to expansion space 8 in the operation of the refrigerator, the working medium flows mainly through the main regenerator 11 while giving up heat to the regenerative filling mass 11a, and partly through the gap 12 forming the auxiliary regenerator, while giving up heat to the metal walls 1a and 2b bounding the gap. When flowing in the reverse direction, the working medium takes up the heat stored in the filling mass 11a and in the walls 1a and 2b.

During operation of the refrigerator a mean pressure drop ΔP occurs through the main regenerator 11 due to flow losses therein, this pressure drop is transmitted to the ends of the gap 12. It has been found that an efficiently operating machine is obtained when the hydraulic diameter d_h of the gap, which is equal to twice the gap width, satisfies the relation:

$$0.4 \times 2.8\sqrt[4]{\frac{s.\eta^2.L}{\rho.\Delta P}} \leqslant d_h \leqslant 1.4 \times 2.8\sqrt[4]{\frac{s.\eta^2.L}{\rho.\Delta P}}$$

where

s=stroke length of the displacer,

η=mean dynamic viscosity of the working medium flowing through the gap during operation of the refrigerator,

L=length of the gap in the axial direction,

ρ=mean density of the working medium flowing through the gap during operation of the refrigerator.

ΔP=mean pressure drop through the main regenerator,

In a cold-gas refrigerator of the above construction, if helium is used as the working medium and the stroke length s of the displacer$=10\times10^{-3}$ m, the mean dynamic viscosity η of the helium$=10^{-5}$ Ns/m^2, the gap length $L=50\times10^{-3}$ m, the mean helium density ρ in the gap$=4.8$ kg/m^3, and the mean pressure drop ΔP through the main regenerator$=0.25$ atm.$=0.25\times10^5$ N/m^2, then

$$2.8\sqrt[4]{\frac{s.\eta^2.L}{\rho.\Delta P}} = 2.8\sqrt[4]{\frac{10\times10^{-3}\times10^{-10}\times50\times10^{-3}}{4.8\times0.25\times10^5}} = 2.8\sqrt[4]{\frac{5\times10^{-14}}{1.2\times10^5}}$$

$$= 2.8\times10^{-5}\sqrt[4]{\frac{50}{1.2}} = 2.8\times2.54\times10^{-5} = 7.1\times10^{-5}\text{m.}$$

The limits of the hydraulic diameter d_h of the gap 12 would then be:

$$2.8\times10^{-5}\ \text{m} \leqslant d_h \leqslant 9.9\times10^{-5}\ \text{m.}$$

WHAT WE CLAIM IS:—

1. A cryogenic refrigerator of the kind described, wherein the communication between the compression and expansion spaces incorporates an auxiliary regenerator which is connected in parallel with the main regenerator and which is formed by an annular gap between the displacer and the cylinder wall cooperating therewith, at least one of the two surfaces of the displacer and said cylinder wall which face each other and bound the gap being formed of a material having a good thermal capacity with respect to the working medium flowing through the gap during operation of the refrigerator, and the hydraulic diameter d_h of the gap satisfying the relation:

$$0.4\times2.8\sqrt[4]{\frac{s.\eta^2.L}{\rho.\Delta P}} \leqslant d_h \leqslant 1.4\times2.8\sqrt[4]{\frac{s.\eta^2.L}{\rho.\Delta P}}$$

where

s=stroke length of the displacer,
η=mean dynamic viscosity of the working medium flowing through the gap during operation of the refrigerator,
L=length of the gap in the axial direction,
ρ=mean density of the working medium flowing through the gap during operation of the refrigerator.
ΔP=mean pressure drop through the main regenerator,

2. A cryogenic refrigerator substantially as herein described with reference to the accompanying drawing.

C. A. CLARK,
Chartered Patent Agent,
Century House,
Shaftesbury Avenue,
London, W.C.2.
Agent for the Applicants.

Printed for Her Majesty's Stationery Office, by the Courier Press, Leamington Spa, 1977
Published by The Patent Office, 25 Southampton Buildings, London, WC2A 1AY, from which copies may be obtained.

1483356 COMPLETE SPECIFICATION

1 SHEET *This drawing is a reproduction of the Original on a reduced scale*